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Cooperative Nuclear Research Station

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A survey of the research in the field of nucleate boiling and film boiling is given. A clear distinction is made between the processes occurring in pool boiling and forced convection boiling. The various kinds of flow patterns associated with the boiling process are discussed in detail. A summary is presented of the most important empirical and theoretical relations for the critical heat flux.

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1. Introduction

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In various experiments that had the purpose of investigating the heat transfer during boiling - especially the transition from nucleate boiling to film boiling (critical thermal stress) - two basic variants must be differentiated (Bibl.31):

a) Heat transfer on heating surfaces in a quasi-static fluid which may be either in the boiling or supercooled state. For this, the term "pool boiling" will be used.

b) Heat transfer during boiling at forced circulation "forced convection

boiling", in which the coolant may pass all phases from a supercooled liquid to a superheated steam. For characterizing the state of the medium, the so-called steam factor x is used, which is expressed by $x = \frac{h - h'}{L}$.

Taking these two variants into consideration, the report will review first the known facts. In this case, the flow distribution, taking place at forced convection, will be discussed in some detail together with the heat transfer mechanism at the respective flow patterns. (As coolant, mainly water is taken into consideration.)

2. Pool Boiling

2.1 General Principles

One of the first and best known investigations in this field was that by S.Nukiyama (Bibl.66). The curve named after him (Fig.1) gives data on the domains of the various forms of evaporation in pool boiling. The slope of this curve, in the domains of interest here - i.e., nucleate boiling and film boiling - depends on the following parameters:

1) type of fluid;

2) system pressure;

3) supercooling;

4) geometry of the test section;

5) surface condition of the heating element; /6

6) material of the heating element;

7) contamination.

Of these factors, we will first consider the two more important parameters of the fluid, namely, pressure and supercooling.

2.2 Influence of Pressure

All published reports (Bibl.1, 47, 60) indicate that the pressure has a distinct influence on the heat transfer in pool boiling, within the zone of nucleate boiling. Figure 2 gives the heat transfer coefficient as a function of the vapor pressure for water, according to Fritz (Bibl.27), based on the equation

$$\alpha = 1.95 \cdot q^{0.72} \cdot p^{0.24} \qquad (1)$$

Mikheyev (Bibl.62) gives the following form for this dependence:

$$\alpha = 3.15 \cdot q^{0.7} \cdot p^{0.15} \qquad (2)$$

The critical thermal stress during pool boiling also depends on the pressure. Figure 3 shows a diagram by Cichelli and Bonilla (Bibl.17), in which the plotting $q_{kr}/p_{cr} = f(p/p_{cr})$ has the advantage of being generally valid for various fluids. The curve was based on measurements with boiling water as well as with several organic liquids in the boiling state.

Of interest are also the experiments by Holt (Bibl.40), in which the Nukiyama curve was rated up to a pressure of 275 atm abs (Fig.3). For comparison purposes, the same diagram gives the curve obtained on the basis of test data by Ye.A.Kazakova (Bibl.45).

2.3 Influence of Supercooling

The experiments by Nukiyama distinctly show a qualitative influence by /7
supercooling on the nucleate boiling as well as on the critical thermal stress. Gunther and Kreith (Bibl.37) mentioned that, with increasing supercooling, the branch of the Nukiyama curve shows a steeper slope in the region of nucleate boiling and that the critical thermal stress increases.

However, an accurate quantitative statement on the influence of supercooling during pool boiling is difficult, since it is impossible to define the reference temperature of the fluid.

As soon as, at a certain supercooling, the heating elements are connected, the liquid near the heating surface heats up rapidly. Thus, the temperature distribution in the vicinity of the test section is not accurately defined and therefore difficult to determine (Bibl.7).

2.4 Theoretical Principles

For the critical thermal stress in pool boiling, various theoretical arguments were developed, of which specifically the following relations are of interest:

a) N.Zuber (Bibl.81) and N.Zuber, M.Tribus, I.W.Westwater (Bibl.82):

$$q_{hcr} = \frac{\pi}{24} L \rho'' \left[\frac{\sigma g (\rho' - \rho'')}{\rho''^2}\right]^{1/4} \cdot \left[\frac{\rho' + \rho''}{\rho''}\right]^{1/2} \qquad (3)$$

for boiling liquids and

$$q_{hcr\,sub} = \frac{\pi}{24} L \rho'' \left[\frac{\sigma g (\rho' - \rho'')}{\rho''^2}\right]^{1/4} + \frac{2 \lambda_l (t_s - t_l)}{\sqrt{\pi \cdot a_l \cdot \tau}} \qquad (4)$$

$$\text{with } \tau = \frac{\pi}{3} \sqrt{2\pi} \left[\frac{\sigma}{g(\rho' - \rho'')}\right]^{1/2} \cdot \left[\frac{\rho''^2}{\sigma g (\rho' - \rho'')}\right]^{1/4}$$

for supercooled liquids.

b) S.Kutadeladze (Bibl.52):

$$q_{hcr} = A \cdot L \cdot \rho''^{1/2} \left[\sigma g (\rho' - \rho'')\right]^{1/4} \qquad (5)$$

where A, depending on the surface condition, has a value of 0.13 - 0.17. /8

For supercooled liquids:

$$q_{hcr_{sub}} = q_{hcr}\left[1 + 0.065\left(\frac{\rho_l}{\rho''}\right)^{0,8} \cdot \frac{c_{pl}(t_s - t_l)}{L}\right] \tag{6}$$

c) W.Rohsenow and P.Griffith (Bibl.72):

$$q_{hcr} = 12.1 \cdot 10^{-3} \cdot L \cdot \rho''\left(\frac{\rho' - \rho''}{\rho''}\right)^{0.6} \tag{7}$$

d) D.H.Labunzov (Bibl.53):

$$q_{hcr} = 188\,\frac{(L\rho''\sigma'\lambda' T_s)^{1/3}}{\nu''^{2/9}}\left(\frac{\rho' - \rho''}{\rho'}\right)^{4/9}, \tag{8}$$

Figure 4 gives the curves for water, based on these formulas, together with the experimental data by Cichelli and Bonilla (Bibl.17) for comparison purposes.

3. Forced Convection Boiling

3.1 General Principles

For forced convection, the same influence factors as in pool boiling are decisive. Beyond this, additional parameters are given by the mass flow (velocity) and, as an extension of the supercooling*, by the steam factor. (The inlet effect will not be further discussed here; only those layouts will be taken into consideration in which the hydrodynamic inflow has ended.) However, the validity of the data obtained in pool boiling can be applied only with certain restrictions to forced convection since the conditions - specifically in the wet-steam domain - are much more complex here.

Therefore, before discussing the conditions in nucleate and film boiling or the critical thermal stress, the various flow patterns, formed by the coolant on passing steam factors from $x < 0$ to $x > 1$, must be considered. /9

* The supercooling can be considered as a negative steam factor.

3.2 Flow Patterns

The flow pattern of the medium on evaporation or cooling is of great importance for evaluating the heat transfer. Since the pipes and channels usually are simultaneously exposed to the flow of both liquid and steam, the processes of motion are much more complex than in a one-phase flow.

Numerous visual observations and attempts to photograph the processes have been made. Generally, two-component mixtures were used as experimental media. Thus, the general concepts on the flow behavior during boiling are based predominantly on observations of air-water or similar two-component mixtures, without heat influx and at low pressures, as well as on processes in horizontal flows or in natural circulation within vertical pipes.

However, investigations on forced convection in vertical direction, with heat influx and especially at elevated and high pressures, are few in number.

No doubt, the majority of this type of work furnished important data on the flow of two-component mixtures; however, it is questionable whether these results can be directly extrapolated to the two-phase flow of a one-component mixture.

In view of the fact that some of these data are basic, we will include the above type of report into our survey. In addition to investigations by other researchers (Bibl.11, 12, 26, 44, 48, 50, 59, 71, 75), the work done by Alves (Bibl.4) and Baker (Bibl.8) is of special importance.

3.2.1 Horizontal Flow of Two-Component Mixtures

The experiments for horizontal flow usually were made with air-water, air-hydrocarbon, air-oil, and natural gas-kerosene mixtures. Alves (Bibl.4) used /10

air-oil and air-water mixtures in his investigations. His observations, together with the data obtained by other investigators, were combined into a nomenclature of flow patterns, illustrated by photographic recordings. Arranged in ascending sequence of gas content, he defined the following types of patterns:

a) Bubble flow, i.e., a flow in which gas bubbles move along the upper part of the pipe, with approximately the same velocity as the liquid.

b) Plug flow, i.e., a flow in which plugs of liquid and gas move along the upper portion of the pipe.

c) Stratified flow, i.e., a flow in which the liquid moves along the bottom of the pipe and the gas flows above this layer, along a smooth liquid-gas interface.

d) Wavy flow, i.e., a flow similar to the stratified flow except that the gas moves at a higher velocity and the boundary layer is disturbed by waves migrating in the direction of flow.

e) Slug flow, i.e., a flow in which the waves are periodically overtaken by the more rapidly moving gas, resulting in the formation of "froth slugs" that pass the pipe at a much higher velocity than the mean velocity of the liquid.

f) Annular flow, i.e., a flow in which the liquid, in the form of a film, flows along the pipewall and the gas flows at high velocity as a central core.

g) Spray flow, i.e., a flow in which most or almost all of the liquid is entrained by the gas as an atomized spray or mist.

Plotting this as the function $w_1 = f(w_g)$ will yield the classification shown in Fig.5 (Bibl.4).

On the basis of data furnished by Jenkins, Gazley, Alves, and Kosterin

(Bibl.42, 30, 4, 48), Baker (Bibl.8) plotted a function $q_{m,g} = f(q_{m,l}/q_{m,g})$ for /11 describing the various flow patterns. Since the data used were obtained mostly with air-water mixtures at atmospheric pressure, Baker introduced correction factors for a generalization of the plotted flow zones, which were to take the physical properties of both liquid and gas into consideration (Fig.6). For this, he used a relation derived by Holmes (Bibl.39):

$$\chi = \left[(\rho_g/\rho_{Air})(\rho_l/\rho_{Water})\right]^{1/2} \text{ and } \psi = (\sigma_{water}/\sigma_l)\left[\mu_l/\mu_{water}\,(\rho_{water}/\rho_l)^2\right]^{1/3}$$

in which the quantities for air and water must be entered at 20° C and atmospheric pressure.

Reports of a more recent date, such as those by Richardson (Bibl.71) and Fohrman (Bibl.25) contain no basically new results.

The plotting of the flow patterns by Baker became one of the most important bases for most of the subsequent investigations.

3.2.2 Vertical Flow of Two-Component Mixtures

Experiments with air-water and two-component mixtures were made also in vertical pipes (Bibl.21, 28, 34, 35, 48, 49, 68, 77).

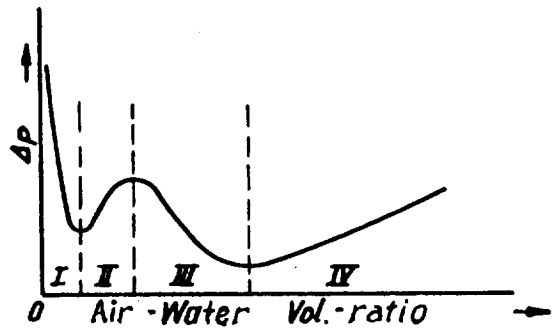
Basically, similar patterns were observed in a vertical flow as they exist in a horizontal pipe flow, except for patterns such as the stratified flow which is characteristic for horizontal two-component motion. Kosterin (Bibl.48) made observations on vertical, inclined, and horizontal pipes of various diameters.

Accordingly, no stratified flow can exist in a vertical pipe. Symmetrical flow patterns were observed only in vertical pipes.

In contrast to numerous authors who founded their description of flow patterns on visual observations and similar bases, Govier et al. (Bibl.34, 35),

according to Radford, Dunn, and Calvert and Williams (Bibl.68, 24, 16), gave /12 a description of the various flow types in connection with their investigations on pressure losses.

The plotted pressure loss curves are subdivided into four segments, characterized by the fact that the pressure drop proceeds monotonically there.



Any fixing of the boundary between two regions is usually determined by a minimum or maximum of the pressure loss curve (see accompanying sketch).

The classification, given by the pressure loss domains, covers the observed flow patterns (Fig.7). In addition to some of their own results, Govier et al. also included the observations by other authors into their classification and listed the data in a Table. These compilations clearly indicate that the boundaries of the individual flow patterns by no means coincide with the boundaries of the pressure loss zones. Rather, it seems that the monotonic slope of the pressure drop undergoes a change as soon as the respective flow pattern is fully formed.

In the opinion of Govier et al., this method offers the advantage of high accuracy in defining the flow patterns.

3.2.3 Natural Two-Phase Flow with Phase Change

The above discussions referred to investigations of two-phase flows, in

which the medium used generally consisted of two components.

An interesting question, however, is the flow behavior if the second phase is produced from the first phase by a phase change, i.e., under heat transfer. Barbet (Bibl.9) made observations as early as 1914 on natural convection in vertical evaporators.

He described the processes as follows: In the lower part of the evaporator, liquid and steam are uniformly distributed. The cocurrent vapor bubbles in- /13
crease in size and number toward the top, finally coalescing, so that the flow will then consist of alternating liquid and vapor slugs. Finally, toward the end of the channel, a gas nucleus forms, while the main liquid continues moving upward along the pipewall as an irregular climbing film flow.

Brooks and Badger (Bibl.14) repeated these experiments and were able to confirm Barbet's observations. Beyond this, just as Stroebe et al. (Bibl.78), they observed an atomized flow with large and fine droplets, i.e., a type of spray flow.

Kirschbaum et al. (Bibl.46) also observed bubble, plug, and spray flow in a vertical evaporator tube, but no annular flow. They encountered no nucleation, which no doubt was due to defects in the apparatus.

Results by Badger, Lewis and Robertson, Reveal, and Dengler (Bibl.6, 57, 69, 23) showed satisfactory agreement with Barbet (Bibl.9).

Lewis and Robertson (Bibl.57) used elevated pressures in their experiments. The flow behaved as in the low-pressure experiments. Dengler (Bibl.23), who worked with premixed steam and water, observed the same flow patterns as in air-water mixtures. However, he noted that bubble and plug flows existed only at very low steam contents.

The totality of these data indicates that the flow patterns of a steam-

water mixture, occurring at natural convection, are similar to those of a two-component mixture.

Conversely, it could be expected that, in the case of forced convection, the influence of factors such as volume velocity and gas content would be considerable.

3.2.4 Forced Two-Phase Flow with Phase Change

Although, in power plant and reactor engineering, forced flow of the coolant (usually at high weight rate of flow) is of considerable importance, only very few data on the flow patterns involved have ever been published. Specifically, no information is available on the behavior of steam-water mixtures at forced convection and high pressures. /14

The concepts on the flow zones of steam-water mixtures, at the above conditions, are practically always based on observations made at lower pressures and predominantly with air and water. An illustrative description of the hydrodynamic processes in vertical heated pipes (Fig.8) was given by Collier (Bibl.19).

Cold water is introduced at the lower end of the heated section. Starting from $x = 0$, the steam component increases until finally 100% steam is flowing in the pipe. During this time, the medium passes heat transfer zones, from the convective heat transfer to water, over supercooled boiling and nucleate boiling etc. to the convective heat transfer to superheated steam. These individual zones are in constant interaction with the fluid-mechanical behavior of the medium.

The flow patterns, to be expected in the individual cases, are shown in the diagram. Accordingly, visible transitions from the one-phase flow of water,

over the two-phase flow of steam + water, to the one-phase flow of steam are clearly defined.

As mentioned also by Collier (Bibl.19), an important item is the change-over from bubble or plug flow to annular flow. Up to this point, the flow behavior of the mixture corresponded to the boiling state. Thereafter, however, the steam component becomes so large that a distinct two-phase flow exists in the form of a liquid film along the wall with a steam core which partly entrains liquid droplets.

However, the boundaries between the individual zones are not clearly defined, since necessary observational data, specifically for high pressures, are absent.

Goldmann et al. (Bibl.32), who plotted the flow patterns as a function of the total volume flow density q_m(kg/m^2/sec) and the steam content x(wt.%) for /15 three different pressures (Fig.9), also had to use the above-described data by Baker (Bibl.8) which were obtained on two-component mixtures.

The plotting indicates that, at pressures of about 100 bar and volume rate of flow of more than 1000 kg/m^2/sec, i.e., under conditions that are of interest for reactor operation, no plug or annular flows were present.

In the model experiment, the medium, under these conditions, flows through the pipe only in the form of a bubble flow, predominantly as a spray flow. Visual or photographic observations in this particular field are extremely rare, since they are difficult to obtain.

To make useful statements, a wide variety of influence factors must be thoroughly studied, such as: history of the medium, channel geometry, volume rate of flow, gravitational and hydrodynamic forces between the phases and within the phases, steam content, density of the phases, etc.

Since a part of these parameters vary constantly as a function of the heat flux and since these variations, in turn, affect the flow behavior, generally valid statements are difficult to make.

One of the most recent investigations in this field, concerned with flow patterns in boiling water, is a study by Tippets (Bibl.79). This author made experiments with forced circulation at a pressure of 70 bar in a rectangular channel. The boiling state extended from $x < 0$ to $x = 0.65$, at heating surface loads up to and including the critical thermal stress.

In connection with a general literature survey, the same author established a terminology of various flow patterns, based in part on the data by Dengler and Kozlov (Bibl.23, 49) and mainly agreeing with the classification by Collier (Bibl.19):

> bubble flow - slug flow - froth flow - annular or climbing film flow - homogeneous (fog) flow.

Tippets' own investigations revealed the dependence of the individual /16 motion pattern on the enthalpy of the liquid or on the steam content and the weight rate of flow.

In the subcooled region and at low steam content, bubble flow and slug flow are clearly defined at low volume rate of flow. However, these stream patterns vanish at a steam content of $x > 0.10$ and velocities of 1 - 2 m/sec.

In the further path within the wet-steam region, an irregular turbulent liquid film, mainly along the unheated wall, was observed. The center of the channel was occupied by a steam flow which entrained liquid in the form of droplets or as a gas-water emulsion.

At velocities of about 2 m/sec, the phenomenon occurred only at very low steam content. At lower velocities, this particular state was reached also at

elevated steam contents.

It is questionable whether a fully formed annular flow or only a transition from slug flow to annular flow was involved here, since the observation no doubt was hindered because of the condensation of liquid on the viewing window.

Basically, however, the observations by Tippets agreed with those by other authors, except for the influence exerted by specially selected factors such as local heating, liquid properties, channel geometry, etc.

Despite the fact that the data obtained in two-component experiments constitute valuable aids, no final conclusions can be drawn as to the flow behavior of a two-phase flow with phase changes, specifically not in the case of forced convection. As mentioned above, a multiplicity of influence factors must be clarified, which are subject to continuous variations due to the influx of heat.

In principle, however, it can be stated that an upward forced convection in a vertical pipe, under supply of heat, will presumably lead to the following flow patterns: /17

bubble flow,

slug flow,

annular flow,

spray or fog flow.

However, no adequate quantitative statement as to the delimitation of the individual regions as a function of the above parameters is possible on the basis of available data.

3.3 Heat Transfer Crisis in the Various Regions

If, under consideration of the statements in the preceding Section, the processes during transition from nucleate boiling to film boiling are to be in-

vestigated, together with the causes responsible for the impairment of heat transfer on reaching the critical thermal load, two domains must be differentiated.

3.3.1 Water Flow with Steam Bubbles

The first domain includes the flow patterns in which the volume percentage of the water is large with respect to that of the steam. Consequently, the water component forms a coherent unit in which the individual bubbles are floating. Within this domain, which includes the flow patterns of "supercooled boiling" and "nucleate boiling", the same transfer mechanism as in pool boiling takes place. The heat is dissipated by the bubbles formed along the heating surface. On increasing the thermal load, the number of active boiling centers as well as the frequency of bubble formation will increase. This causes an increase in density of the bubbles, which reaches such a high value that the individual bubbles coalesce to larger units until finally a closed steam film is formed, which isolates the heating surface from the core flow. The term "film boiling" should be reserved exclusively for the type of boiling produced in this particular manner and leading to an abrupt increase in the pipewall temperature; only in this particular case should expressions such as "critical thermal stress" (departure from nucleate boiling) be used. /18

3.3.2 Steam Flow with Water Droplets

The second domain comprises the portion of the wet-steam region in which the coherent water flow has been disrupted and in which a steam flow, interspersed with liquid, is present. This particular domain includes the flow patterns known as "annular flow" and "spray flow". According to Collier (Bibl.20),

the heat in the case of an annular flow is transferred exclusively by heat conduction through the liquid film formed along the wall, accompanied by evaporation along the inner surface of this film. The film is so thin that, during the described heat transfer process, the excess temperature of the wall is below the temperature required for activating the boiling process, so that no bubbles are able to form. Conversely, the liquid film is constantly replenished from the core flow with its entrained droplets; this film remains stable as long as sufficient droplets are available for replacing the evaporated liquid. However, as soon as - at increasing thermal stress - the rate of evaporation becomes equal to the rate of replacement, a degradation of the liquid film begins. This so-called "drying" of the film with the resultant more or less abrupt increase in the temperature of the heating surface, is considered by most authors (Bibl.16, 32) as a crisis, in which a critical thermal stress is usually described. This raises the question whether the impairment of the heat transfer, due to the destruction of the liquid film on the heating surface, actually represents a crisis phenomenon or whether it might not be a process which always occurs on any complete evaporation of water in forced convection (Bibl.83).

The same statements apply to spray flow. Here, the heat transfer - at a lower load on the heating surface - is still satisfactory since the liquid droplets contribute to the cooling. With increasing thermal stress, the steam along the heating surface is so greatly superheated that the liquid droplets are no longer able to penetrate to the wall for cooling it, but evaporate before reaching the wall, thus contributing to the increase in temperature of the heating surface.

Accordingly, within the above-described domain of the wet-steam region, /19 one cannot speak of a true nucleate boiling or of a film boiling, in the sense

of the processes described in Section 3.3.1. Rather, the expression "impaired heat transfer in the wet-steam region" should be used.

3.3.3 Delimitation of the Regions

For a rough estimate of the two domains, the following theory can be established, in agreement with Zwickler (Bibl.83): Imagining a cooling channel filled with the densest possible cubic bubble packing (spherical), the maximum ratio of steam volume to total volume would be $\frac{\pi}{6}$. Tetrahedral or octahedral arrangements would increase the possible bubble volume, but this seems a somewhat exaggerated critical case assumption. As a possible simplification, the term "water flow with steam bubbles" could be used for describing a flow with a steam volume content up to 0.5, while the term "steam flow with water content" is useful for describing a flow with a superposed steam volume component.

In Fig.10, the steam component by weight x is plotted against the pressure, for water at this particular steam volume component.

In this case, the relation

$$\alpha_v = \frac{1}{1 + S \frac{V'}{V''} \cdot \frac{1-x}{x}} \qquad (9)$$

is valid. The solid curve in the diagram represents an effective slip at q_s = = 1000 kg/m²/sec (Bibl.58, 74), while the broken line applies to the simplified case S = 1.

Obviously, especially at low to moderate pressures, the region of steam flow with water content predominates. Since the steam component by weight is proportional to the heating of the medium and thus, in the case of a uniformly heated pipe, also proportional to the pipe length, the region of bubble flow in the wet-steam zone most likely will extend over only a short distance, in most

experiments with water.

3.4 Experimental Results and General Formulas /20

3.4.1 Nucleate Boiling

Within the domain of nucleate boiling, the influences of the various parameters have the same tendency as in pool boiling. Published reports (Bibl.13, 27, 47) indicate that, in supercooled boiling as well as in normal boiling, the influence of the velocity is masked by the elementary process of nucleate boiling.

In considering the influence of supercooling, the selection of the reference temperature is decisive. If the stress on the heating surface is plotted against the temperature difference $t_w - t_b$ (Fig.11), a distinct dependence on the supercooling will be observed (see also Section 2.3). However, if the temperature difference $t_w - t_s$ is used as abscissa, all curves will coincide within the region of nucleate boiling (Fig.12). As already indicated by McAdams (Bibl.60), this shows that, for interpreting the results, only the temperature difference $t_w - t_s$ is decisive and that the supercooling also has no influence on the nucleate boiling.

Next to the McAdams curve, which thus applies to various supercoolings and velocities at a pressure of about 2 - 6 bar, we entered the straight line for 5 bar

$$q_h = 2.55 \cdot e^{P/15.5} (t_W - t_S)^4 \qquad (10)$$

according to the equation derived by Jens and Lottes (Bibl.43). For better illustration, the relations by Fritz and Mikheyev (see Section 2.2) for atmospheric pressure are also plotted.

Since, according to the above statements, the influence of velocity and

supercooling is swamped in the region of nucleate boiling in the case of forced convection, the dependence on the pressure would have to be the same as in pool boiling. In fact, a satisfactory agreement of the various test data of both variants was found (Bibl.13, 47) so that one could say that the curves in Fig.12, for a given liquid, are exclusively determined by the pressure, dis- /21 regarding external influences such as condition of the heating surface or contamination of the liquid.

3.4.2 Critical Thermal Stress

For determining the critical thermal stress in heating elements exposed to the flow and for investigating the influences of the individual parameters, three different methods are in question according to Collier (Bibl.19).

3.4.2.1 Theoretical Method

Several authors (Bibl.36, 55, 70), as in the case of pool boiling, established theoretical methods for correlating the hydrodynamic states and heat transfer conditions, present in a cooled heating element, with the conditions for critical thermal stress. Griffith (Bibl.36) derived an equation which is composed of several summands that allow for the influence of velocity and the influence of supercooling:

$$\frac{q_{hcr}}{(h''-h_b)\,\rho''\left[\frac{\rho'-\rho''}{M'}\,g\left(\frac{\lambda'}{\rho'\,c'_p}\right)^2\right]^{0.33}\cdot F}\cdot f\left(\frac{p}{p_{cr}}\right) \qquad (11)$$

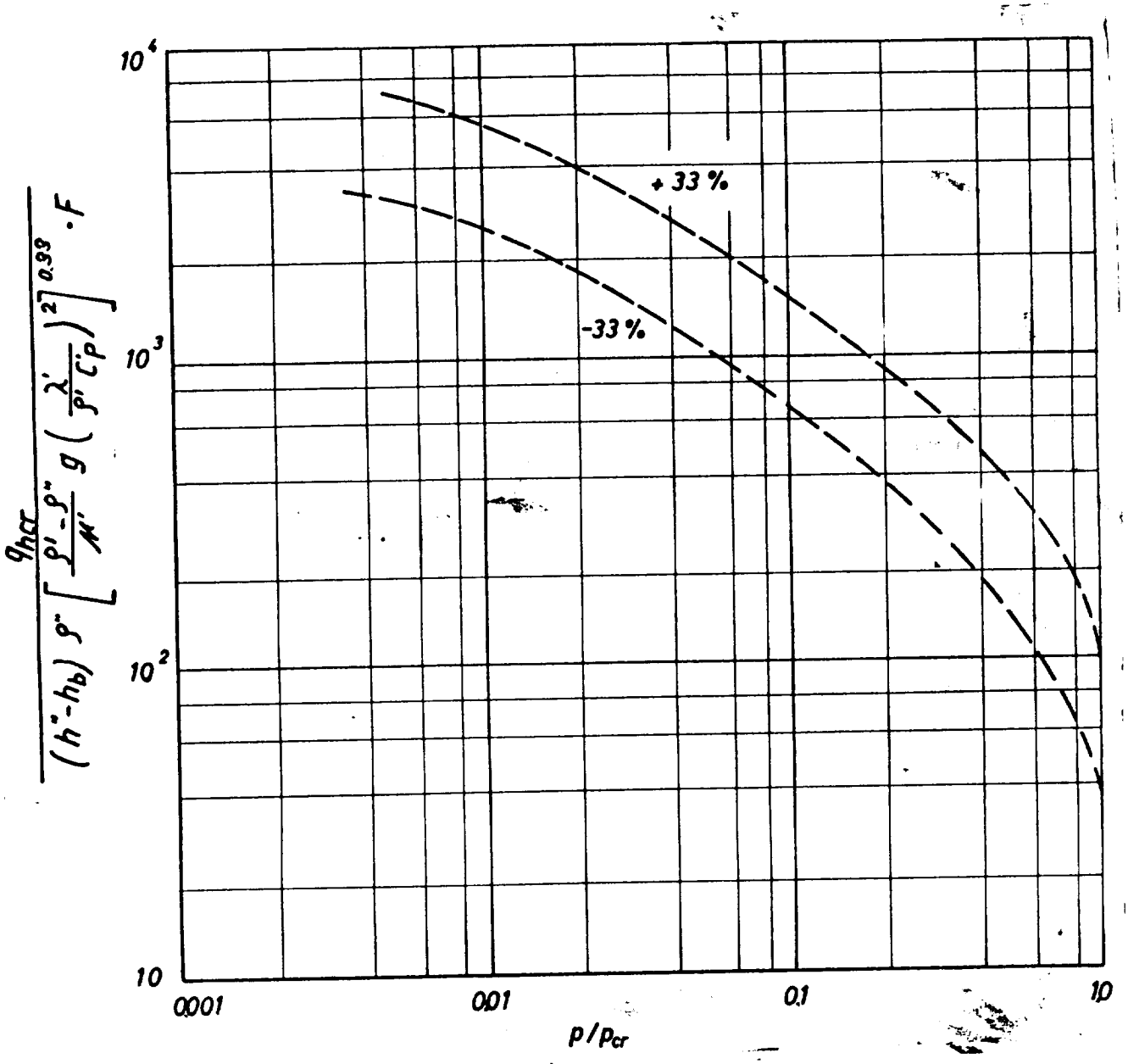
with

$$F = 1+\frac{w\cdot d_h\cdot\rho'}{M'\cdot 10^6}+0.014\,\frac{\rho'\cdot c'_p\,(t_s-t_b)}{\rho''\,L}+0.5\cdot 10^{-3}\left[\left(\frac{w\cdot d_h\cdot\rho'}{M'}\right)\frac{\rho'\cdot c'_p\cdot(t_s-t_b)}{\rho''\,L}\right]^{0.5} \qquad (12)$$

This expression is valid also for the wet-steam region, in which case the last

term and the term before the last become zero.

The function according to eq.(11) was determined on the basis of numerous available experimental data and is plotted in the accompanying diagram. The



curves are limiting curves that cover a scattering range of ±33%. Their validity range is

$$0 \leq \Delta t_{sub} \leq 140^{\circ}C$$
$$0 \leq W \leq 33.5 \text{ m/sec}$$
$$0 \leq X \leq 70\ \%$$

Levy (Bibl.55) used Zuber's theory (Bibl.81, 82) in supplementing eq.(4) /22 for subcooled pool boiling with another term for forced convection, of the form

$$q_{hcr} = \alpha_L (t_w - t_s) + \alpha_L (t_s - t_b) . \tag{13}$$

Here,

$$\alpha_L = 0.023 \frac{\lambda_L}{d_h} \cdot Re_L^{0.8} \cdot Pr_L^{0.33}$$

and, according to eq.(10), /23

$$(t_w - t_s) = \frac{25}{e^{p/62}} \left(\frac{q_h}{10^6}\right)^{1/4}$$

Addition of eqs.(4) and (13) will then yield q_{hcr} at forced convection for a supercooled liquid, in which case $t_w - t_s$ must be iteratively determined. This relation was further developed by Levy (Bibl.55, 56) for the wet-steam region, by adding another term for the material exchange in the two-phase flow.

3.4.2.2 Empirical Method

This method is based on experiments made in various laboratories under entirely definite test conditions. The results were used for deriving equations that permit a prediction as to the maximum heat flux, within certain limits. Jens and Lottes (Bibl.43), gave the following relation for pipes with water flow at subcooling:

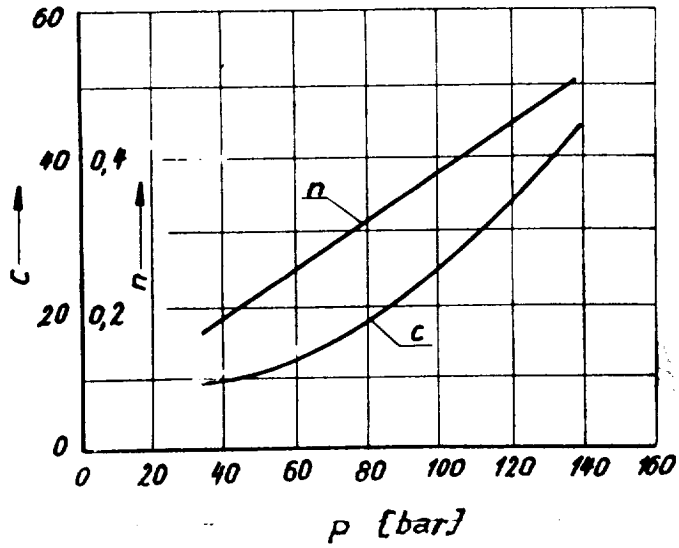
$$q_{hcr} = C \cdot \left(\frac{q_m}{10^6}\right)^n \Delta t_{sub}^{0.22} \cdot 10^6 \tag{14}$$

The validity range is

$$35 \leq p \leq 140 \text{ bar}$$
$$1500 \leq q_m \leq 9000 \text{ kg/m}^2\text{sec}$$
$$2 \leq \Delta t_{sub} \leq 90^\circ\text{C}$$
$$3.6 \leq d_i \leq 5.8 \text{ mm}$$

The values for C and n are plotted as a function of the pressure, in the

accompanying diagram.



McAdams (Bibl.61) used annular channels with water flow in his experiments, of the dimensions /24

$$d_i = 6.2 \text{ mm}; \; d_o = 11, 18.5, \text{ and } 19 \text{ mm}$$

yielding the relation

$$q_{hcr} = (1.87 \cdot 10^6 + 4.05 \cdot 10^4 \, \Delta t_{sub}) w^{0.33} \qquad (15)$$

for

$$2 \leq p \leq 6 \text{ bar}$$
$$0.3 \leq w \leq 3.6 \text{ m/sec}$$
$$11 \leq \Delta t_{sub} \leq 55^\circ C$$

Additional empirical formulas for subcooled water were derived by Gunther (Bibl.38)

$$q_{hcr} = 7.2 \cdot 10^4 \cdot w^{0.5} \, \Delta t_{sub} \qquad (16)$$

for rectangular channels and the validity range

$$1.3 \leq p \leq 11 \text{ bar}$$
$$1.5 \leq w \leq 12.0 \text{ m/sec}$$
$$11 \leq \Delta t_{sub} \leq 155^\circ C$$

as well as by Buchberg (Bibl.2):

$$q_{hcr} = 5 \cdot 10^4 \cdot q_m^{0.5} \cdot \Delta t_{sub}^{0.2} \tag{17}$$

Averin and Krushilin (Bibl.5) made their experiments on annular orifices of 22/6 mm diameter, in water flow; however, the tests were made at $x \approx 0$, yielding the empirical formula

$$q_{hcr} = 1.95 \cdot 10^6 \cdot w^{0.5} \cdot p^{0.33} \tag{18}$$

for

$$1 \leq p \leq 9 \text{ bar}$$
$$0.850 \leq w \leq 5.5 \text{ m/sec}$$

For the wet-steam region, tests were made under various experimental conditions, specifically by R.A.DeBortoli et al. (Bibl.22) as well as by Cise (Bibl.76). As test sections, DeBortoli used tubes of 2 - 8 mm diameter as well as rectangular channels with cross sections of 25 × 2.5 mm and 25 × 1.27 mm, respectively. The equations for the critical thermal stress are as follows: /25

$$q_{hcr} = 72.5 \cdot 10^{20} \cdot h_b^{-2.5} \left(1 + \frac{q_m}{1.36 \cdot 10^4}\right)^2 \cdot e^{-0.0012\, l/d} \tag{19}$$

valid for pipes at $0.27 \times 10^3 \leq q_m \leq 11 \times 10^3$ kg/m²/sec,

$$q_{hcr} = 96 \cdot 10^{20} h_b^{-2.5} \left(1 + \frac{q_m}{136 \cdot 10^4}\right)^2 \cdot e^{-0.0012\, l/d_r} \tag{20}$$

valid for rectangular channels at $2.2 \times 10^3 \leq q_m \leq 6.8 \times 10^3$ kg/m²/sec,

$$q_{hcr} = 130 \cdot 10^{20} \cdot h_b^{-2.5} \cdot e^{-0.0012\, l/d_r} \tag{21}$$

valid for rectangular channels at $0.27 \times 10^3 \leq q_m \leq 2.2 \times 10^3$ kg/m²/sec.

For all three relations, the following limits are applicable:

Pressure $25 \leq p \leq 150$ bar

Enthalpy $1.16 \times 10^6 \leq h \leq 2.32 \times 10^6$ J/kg.

In addition, for pipes: $21 \leq l/d \leq 365$

for channels: $59 \leq l/d_r \leq 468$.

The above author suggests, for the thermomechanical layout of a given project, to multiply the value for q_{hcr} obtained from the above formula by 0.65, to obtain a sufficiently safe value.

Similarly, Cise (Bibl.76) made a large number of tests in the wet-steam region, on tubes of 3 - 10 mm inner diameter with water flow, at a pressure of predominantly 70 bar, a value of x of 15 - 80%, and a mass rate of flow of /26 1000 - 4000 kg/m²/sec. The results give a satisfactory view over the critical heat transfer conditions in the wet-steam region, within the given limits.

Miropolskiy and Shitsman (Bibl.63) constructed an alignment chart for determining q_{hcr} in tubes and annular orifices in water flow, within the wet-steam region (up to x = 0.6); Fig.13.

3.4.2.3 Statistical Method

This method is based on a statistical evaluation of numerous test data.

Jacobs and Merril (Bibl.41), for the case of water flow in pipes at supercooling, derived the following polynomial, replacing the steam factor by the inlet temperature and by the pipe length as reference values.

$$\begin{aligned} q_{hcr} = {} & 2.0245.10^{6} - 2.3296.10^{4}t + 4.3354.10^{4}p + 2.9617.10^{3}q_m \\ & -5.0828.10^{7}l + 3.2334.10^{6}d + 1.0521.10^{2}t^2 - 3.6160.10^{2}p^2 \\ & -0.2463\,q_m^2 + 4.6857.10^{7}l^2 + 0.8180.10^{2}.pt - 8.3783.tq_m \\ & -1.6968.10^{5}d^2 - 0.2314t^3 + 3.7275.pq_m + 3.9739.10^{4}pl \\ & -5.7503.10^{3}pd - 1.0470.10^{3}q_m l + 0.9683.10^{2}.q_m d + 8.4899.10^{4}ld \\ & +0.7119.p^3 + 0.8126.10^{-5}q_m^3 - 4.3031.10^{6}l^3 + 4.2708.10^{-4}tq_m^2 \end{aligned} \qquad (22)$$

This is valid for

$$34 \leq p \leq 185 \text{ bar}$$
$$280 \leq q_m \leq 10500 \text{ kg/m}^2\text{sec}$$
$$25 \leq t \leq 325 \text{ °C}$$
$$2 \leq d \leq 8 \text{ mm}$$
$$0.15 \leq l \leq 0.7 \text{ m} .$$

Bell (Bibl.10) developed a similar method for water flow in rectangular channels, at a pressure of 140 bar and a mass flow rate of 270 - 6800 kg/m^2/sec, yielding the equation

$$q_{hcr} = a \cdot 10^6 \left(b + \frac{q_m}{10^3}\right)^c \left(\frac{h'' - h_b}{d \cdot 10^6}\right)^{e(q_m/10^3)^f} . \quad (23)$$

For the constants a to f, the following values are applicable: /27

	$1250 \cdot 10^3 \leq h_b \leq 1562 \cdot 10^3$	$1563 \cdot 10^3 \leq h_b \leq 2300 \cdot 10^3$
a	0.775	2.785
b	3.51	1.61
c	1.81	1.44
d	2.32	2.32
e	1.56	1.45
f	0.406	0.474

3.5 Dependence of the Critical Thermal Stress on Various Parameters

Despite the multiplicity of the formulas, given in the preceding Sections, it is difficult to establish a generally valid relation that would permit a quantitative prediction of the critical thermal stress.

The available test data merely furnish statements as to the qualitative slope of q_{hcr} as a function of the respective parameters, namely, pressure,

mass flow, and steam factor (Bibl.20, 73, 79). The influence of pressure, in the case of forced convection, is adequately defined only in the pressure range above 40 bar, which clearly indicates that q_{hcr} decreases in this region with increasing pressure.

Beyond this, it can be assumed that the curve $q_{hcr}/p_{cr} = f(p/p_{cr})$ has a similar slope as in pool boiling, except that the maximum is shifted in the direction of lower pressures and that the absolute values depend on the respective state of the coolant (Bibl.67). Numerous, partly contradictory, theories on the influence of the volume flow density on variation in the vapor coefficient have been established during the course of the experimental periods.

Figure 14 gives a schematic diagram according to Collier (Bibl.20); the arrows point in the direction of increasing mass flow density. The upper diagram (Fig.14a) represents the conventional opinion of various authors, namely that q_{hcr} increases with increasing mass flux over the entire region and /28 monotonically decreases at higher steam content.

Pexton (Bibl.67) and also Bell (Bibl.10) were of the opinion that, starting from a certain steam factor, an inversion of the influence of the volume flow density begins. This phenomenon is plotted in Fig.14b for a pressure of 140 bar, in which case the inversion takes place at a steam content of about 15%. This process is explained by the fact that, in the field of the annular flow, the water film formed on the pipewall becomes thinner at higher mass flux (Bibl.18).

Collier went a step further and gave the representation plotted in Fig.14c for pressures below 140 bar. At a high volume rate of flow, the critical thermal stress decreases monotonically with increasing steam factor, as in the other upper diagrams. However, with decreasing volume rate of flow, the curve ascends again at a steam factor of 15 - 30%, reaches a maximum, and then descends mono-

tonically. This inversion is more pronounced and extends over a range of the steam factor that is greater the lower the mass flux. Simultaneously, an inversion of the influence of the volume flow density takes place. The presence of a maximum in the family of curves q_{hcr} versus the steam factor x leads to the fact that the so-called burnout point will be located at the center of the pipe instead of at the end, as would have been expected. This fact has resulted in erroneous data in numerous experimental setups for determining the critical thermal stress, especially when using a "burnout detector".

The plotting in Fig.14c was confirmed by results obtained in various experimental laboratories (Bibl.76, 80); however, the available test data are insufficient for defining the accurate limits of the described inversions.

At a pressure above 140 bar, the family of curves presumably will again show a monotonic slope in accordance with Fig.14a, as indicated by previous experimental series at very high pressures (Bibl.64). In addition, it is of importance that, in determining the critical thermal stress, the experimental /29 setup must be free of pulsations, since it was found that, in the case of an unstable flow, inversions may occur within the slope of q_{hcr}, as a function of x and p (Bibl.3).

List of Symbols /30

a	(m^2/sec)	=	Temperature coefficient
c_p	(J/kg/°)	=	Specific heat at constant pressure
d	(m; mm)	=	Diameter
d_h	(m)	=	Hydraulic diameter
g	(m/sec^2)	=	Gravity
h	(J/kg)	=	Enthalpy

ℓ	(m)	=	Length
L	(J/kg)	=	Heat of vaporization
p	(bar)	=	Pressure
Pr		=	Prandtl number
q_h	(w/m^2)	=	Stress on heating surface, heat flux density
q_m	($kg/m^2/sec$)	=	Volume rate of flow, mass flux
Re		=	Reynolds number
S		=	Slip (w_g/w_l)
t	(^oC)	=	Temperature
T	(^oK)	=	Temperature
v	(m^3/kg)	=	Specific volume
w	(m/sec)	=	Velocity
x	(wt.%)	=	Steam content, steam factor
α	($w/m^2/^o$)	=	Heat transfer coefficient
α_v	(vol.%)	=	Steam content
λ	($w/m/^o$)	=	Heat conductivity
μ	(kg/m/sec)	=	Dynamic viscosity
ν	(m^2/sec)	=	Kinematic viscosity /31
ρ	(kg/m^3)	=	Density
σ	(N/m)	=	Surface tension
τ	(sec)	=	Period

Indices /32

o	=	Outer
b	=	Bulk
cr	=	Critical

g = Gas (steam)

i = Inner

ℓ = Liquid

m = Mass

r = Distance between the heating surfaces in a rectangular channel

s = Saturation

sub = Subcooling

v = Volume (void)

w = Wall

The indices ' and " characterize the variables of state for water and steam in the saturated state.

Translation for Foreign Bibliography Items /33

7. Unpublished Experiments.

13. Calculation of Heat Transfer at Fully Formed Nucleate Boiling.

27. Principles of Heat Transfer in the Evaporation of Liquids.

31. Research in the Field of Film Boiling.

46. Heat Transfer in Vertical Evaporator Tubes.

47. Heat Transfer on Boiling.

53. Generalized Relation for the Critical Heat Flow Density on Boiling of Liquids, at Free Convection.

62. Principles of Heat Transfer.

64. Internal Heat Transfer in Forced Convection Tanks (Film Boiling).

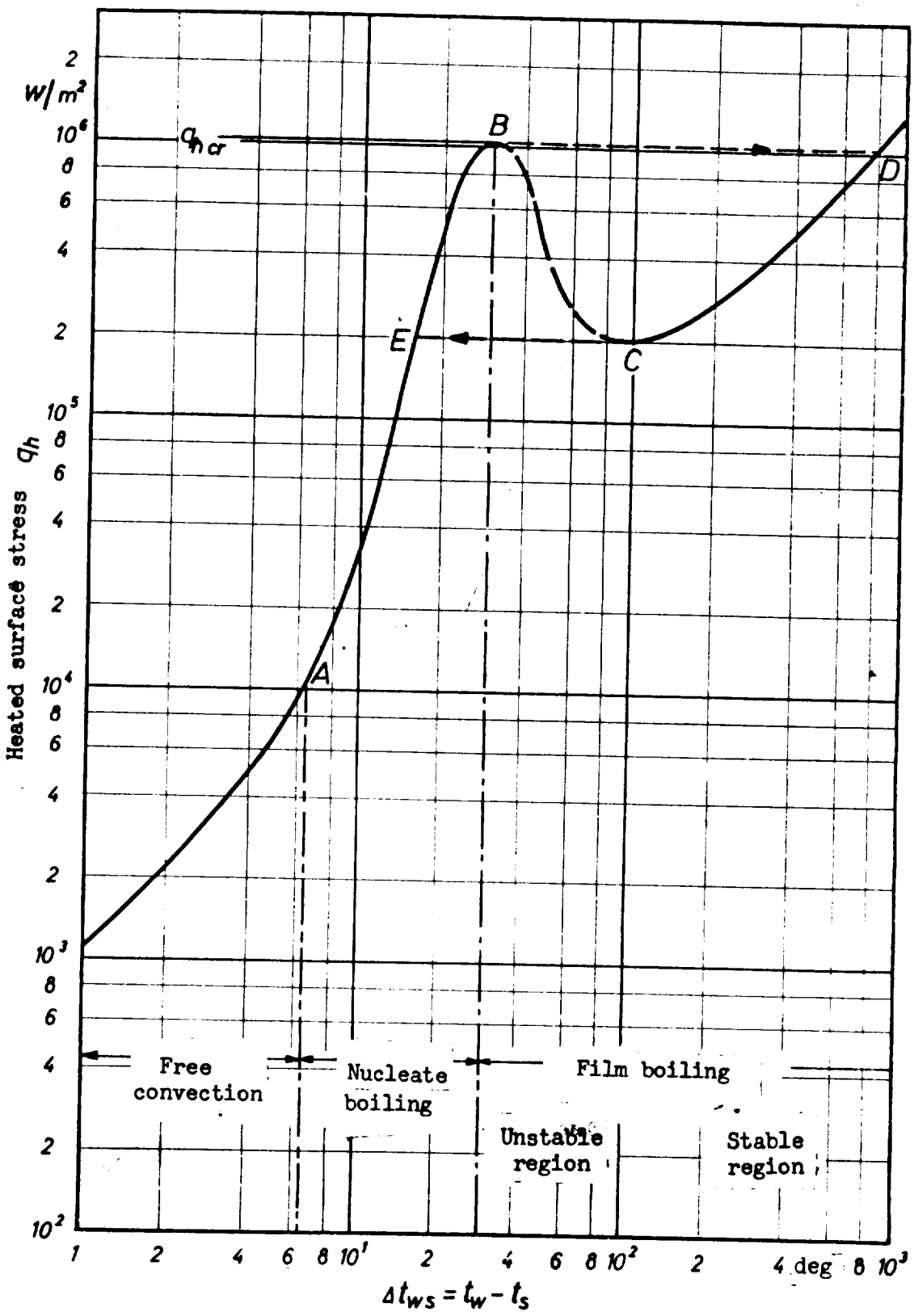
73. Thermomechanical Investigations on Highly Stressed Boiler Heating Surfaces.

74. Investigations on Density Distribution, Water and Steam Velocity, and Friction Pressure Drop in Vertical and Horizontal Boiler Risers.

83. Studies on the Problem of Light-Water Moderated Forced Convection Reactors, with Superheating at Subcritical Pressure.

Illustrations /45

1. General Boiling Curve according to Nukiyama (Bibl.66) for Pool Boiling at Atmospheric Pressure.
2. Heat Transfer for Water at Various Pressures (Bibl.27).
3. Critical Heating Surface Stresses as a Function of Pressure (Experimental Results).
4. Critical Heating Surface Stresses as a Function of Pressure (Theoretical Relations).
5. Flow Zones according to Alves (Bibl.4).
6. Generalized Representation of the Flow Regions (Bibl.8).
7. Representation of Pressure Loss and Flow Regions, according to Investigations by Various Authors (Bibl.34).
8. Two-Phase Flow Regions according to Collier (Bibl.19).
9. Flow Regions for Steam-Water at Various Pressures (Bibl.32).
10. Maximum x-Values for Bubble Flow as a Function of Pressure (Bibl.83).
11. Influence of Subcooling and Velocity on the Heat Transfer Conditions in the Evaporation of Supercooled Water (q_h as a Function of $t_w - t_b$).
12. Comparison of the Slope of Heat Transfer to Water, in the Region of Nucleate Boiling (q_h as a Function of $t_w - t_s$). /46
13. Nomogram for Determining the Critical Thermal Stress in Pipes and Annular Orifices ($l/d_h > 100$).
14. Slope of Critical Thermal Stress, as a Function of the Steam Factor.


W/m²
Heated surface stress q_h
$q_{h\,cr}$
A
B
C
D
E
Free convection
Nucleate boiling
Film boiling
Unstable region
Stable region
deg
$\Delta t_{ws} = t_w - t_s$

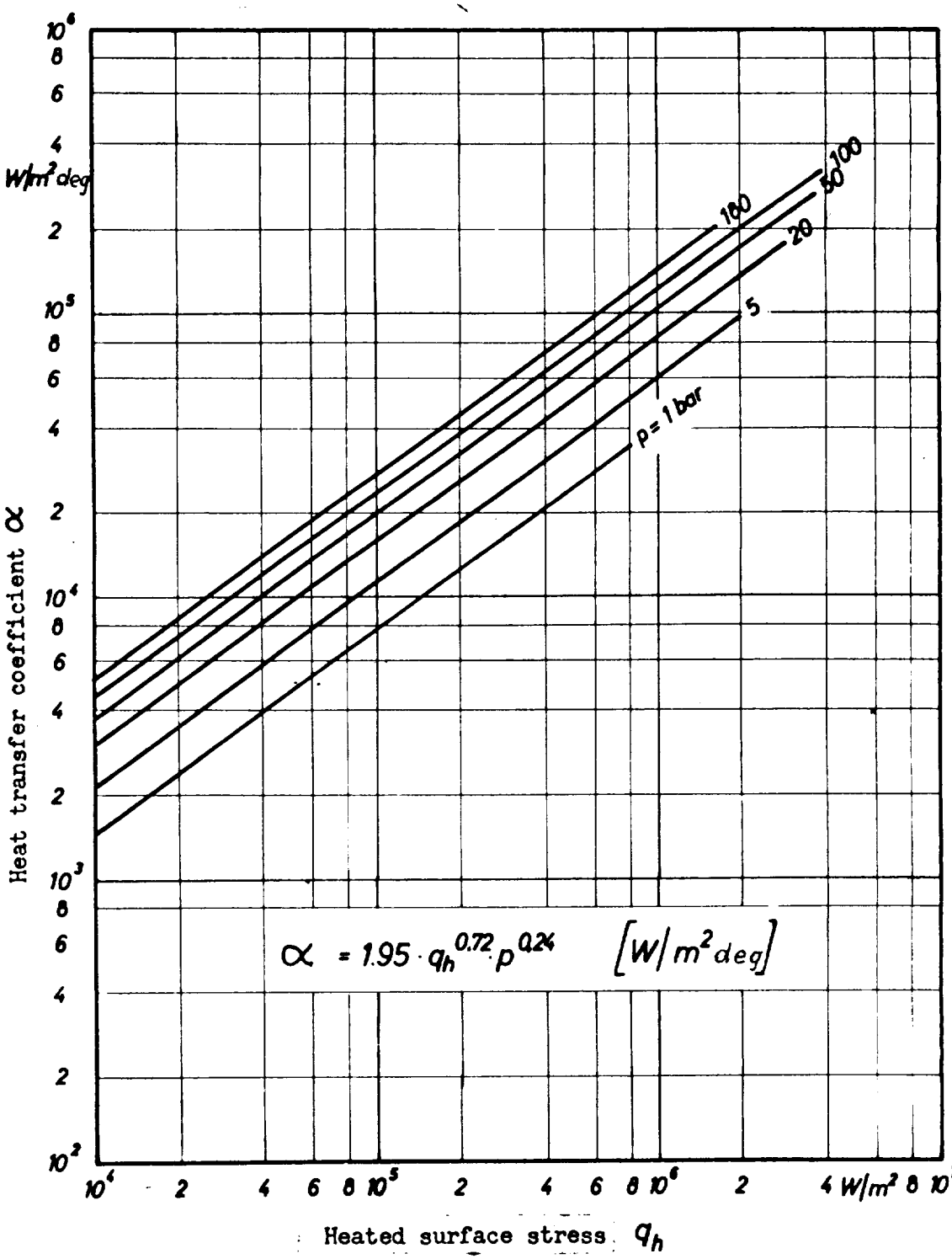


Fig.2 Heat Transfer for Water at Various Pressures (Bibl.27)

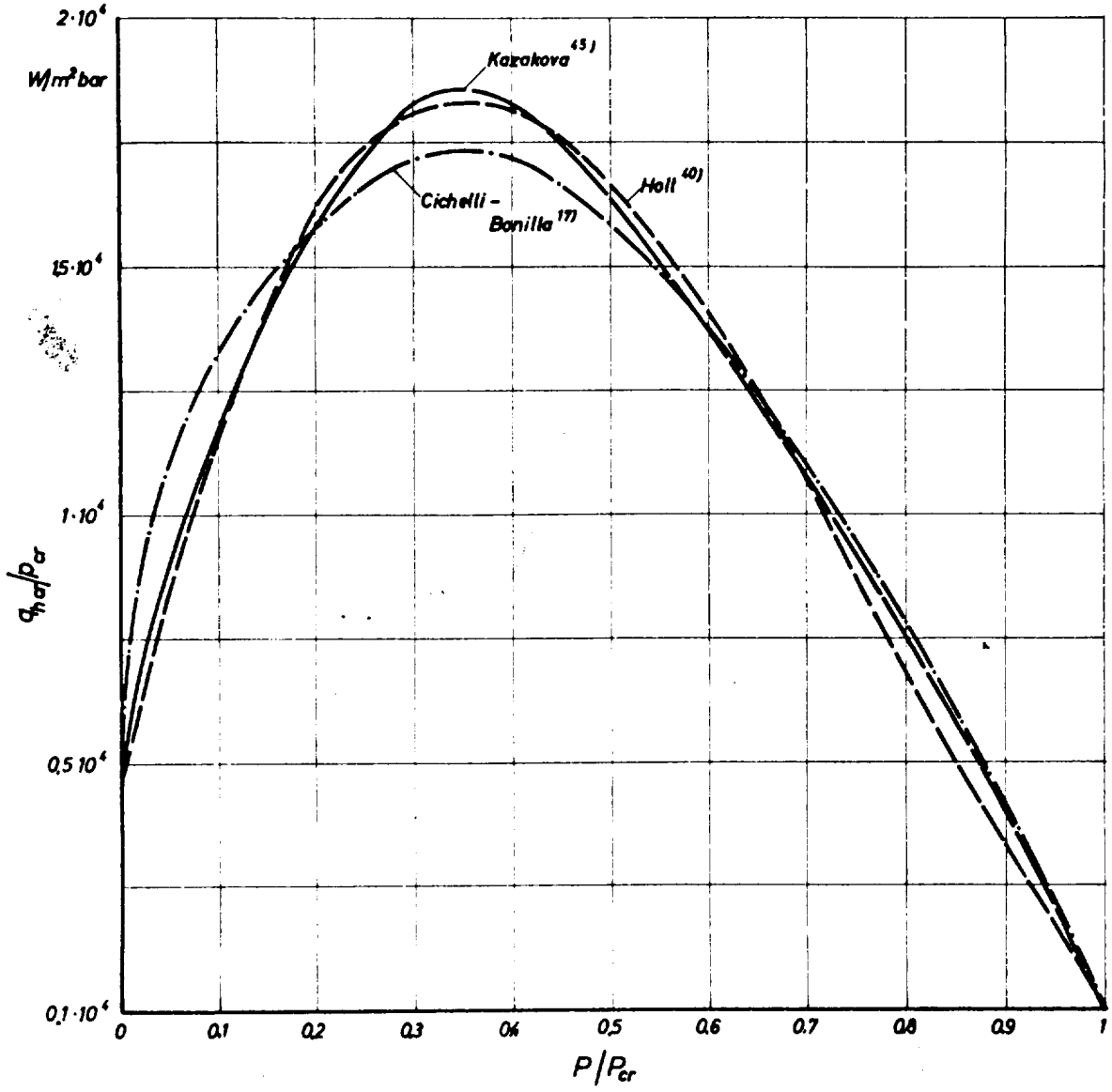


Fig.3 Critical Heating Surface Stresses as a Function of Pressure (Experimental Results)

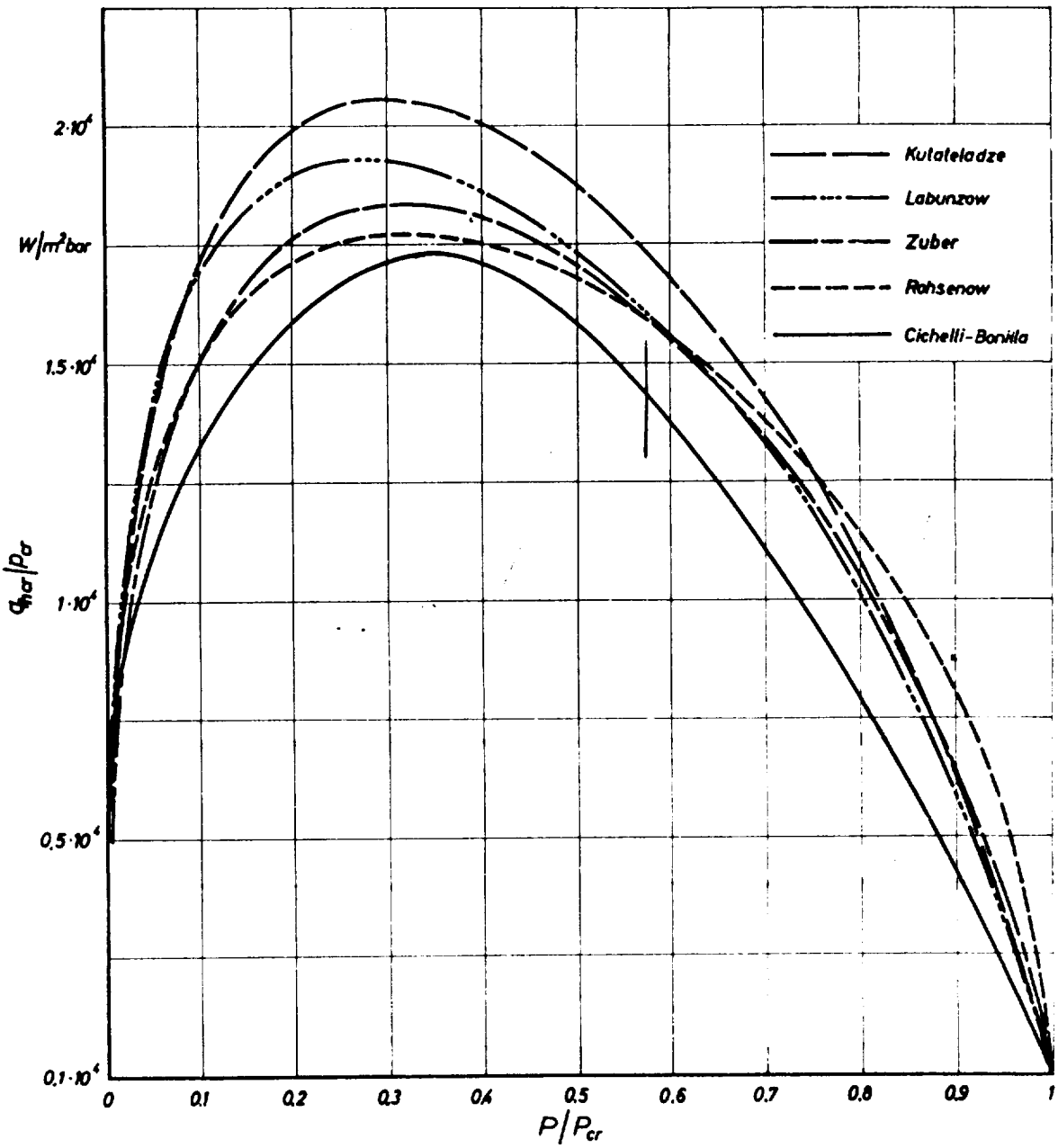


Fig.4 Critical Heating Surface Stresses as a Function of Pressure (Theoretical Relations)

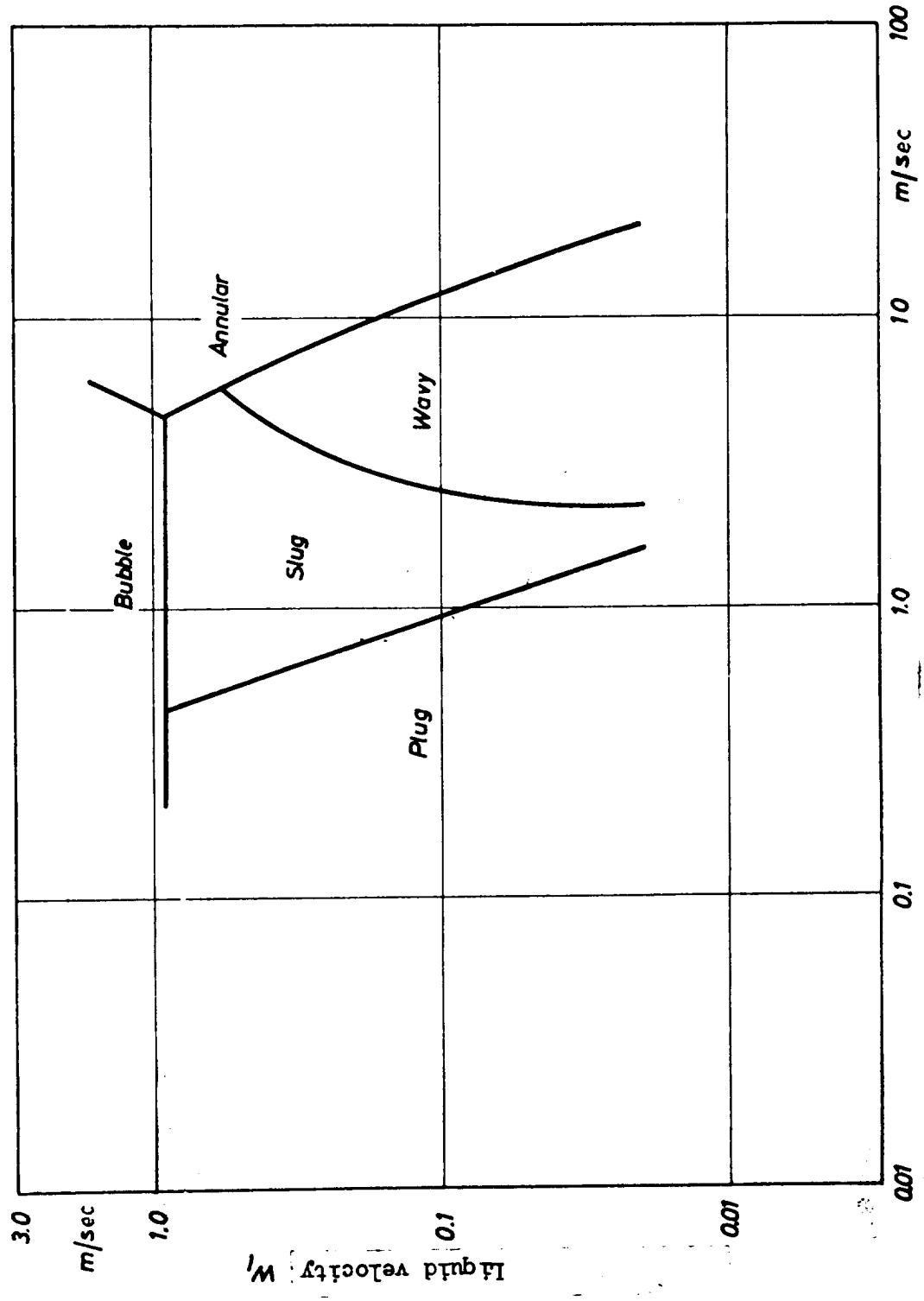


Fig.5 Flow Zones according to Alves (Bibl.4)

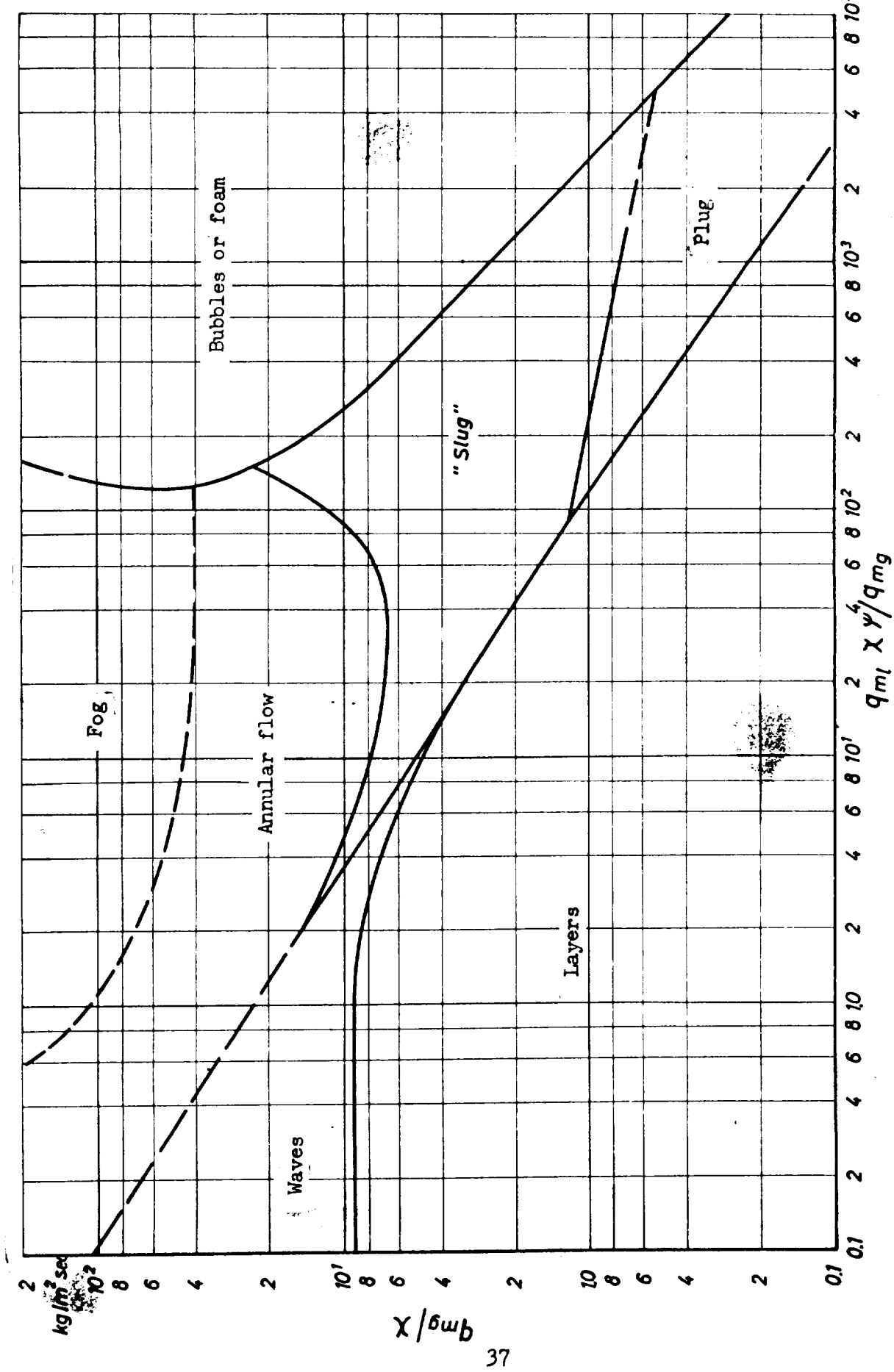


Fig.6 Generalized Representation of the Flow Regions (Bibl.8)

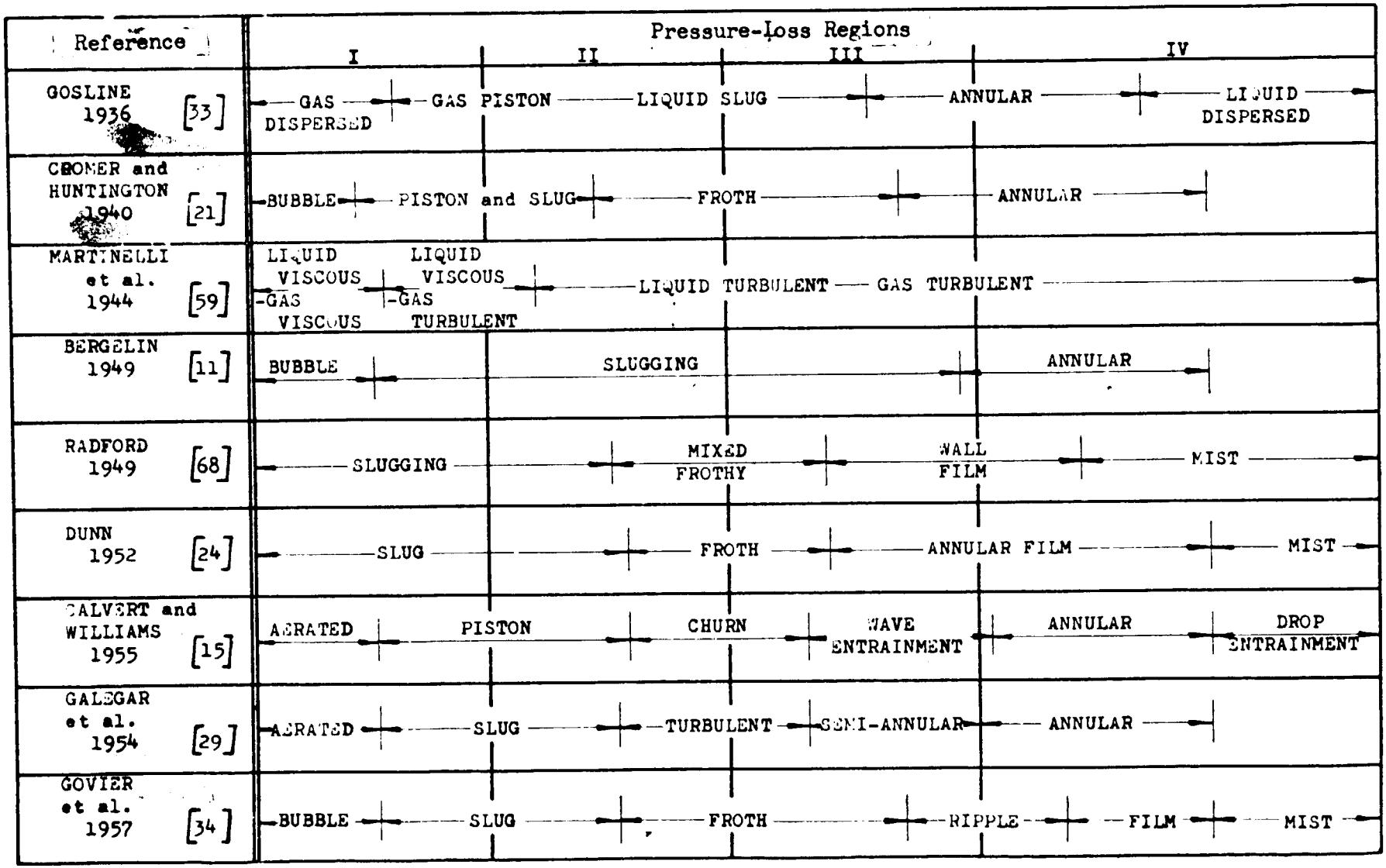


Fig.7 Representation of Pressure Loss and Flow Regions, according to Investigations by Various Authors (Bibl.34)

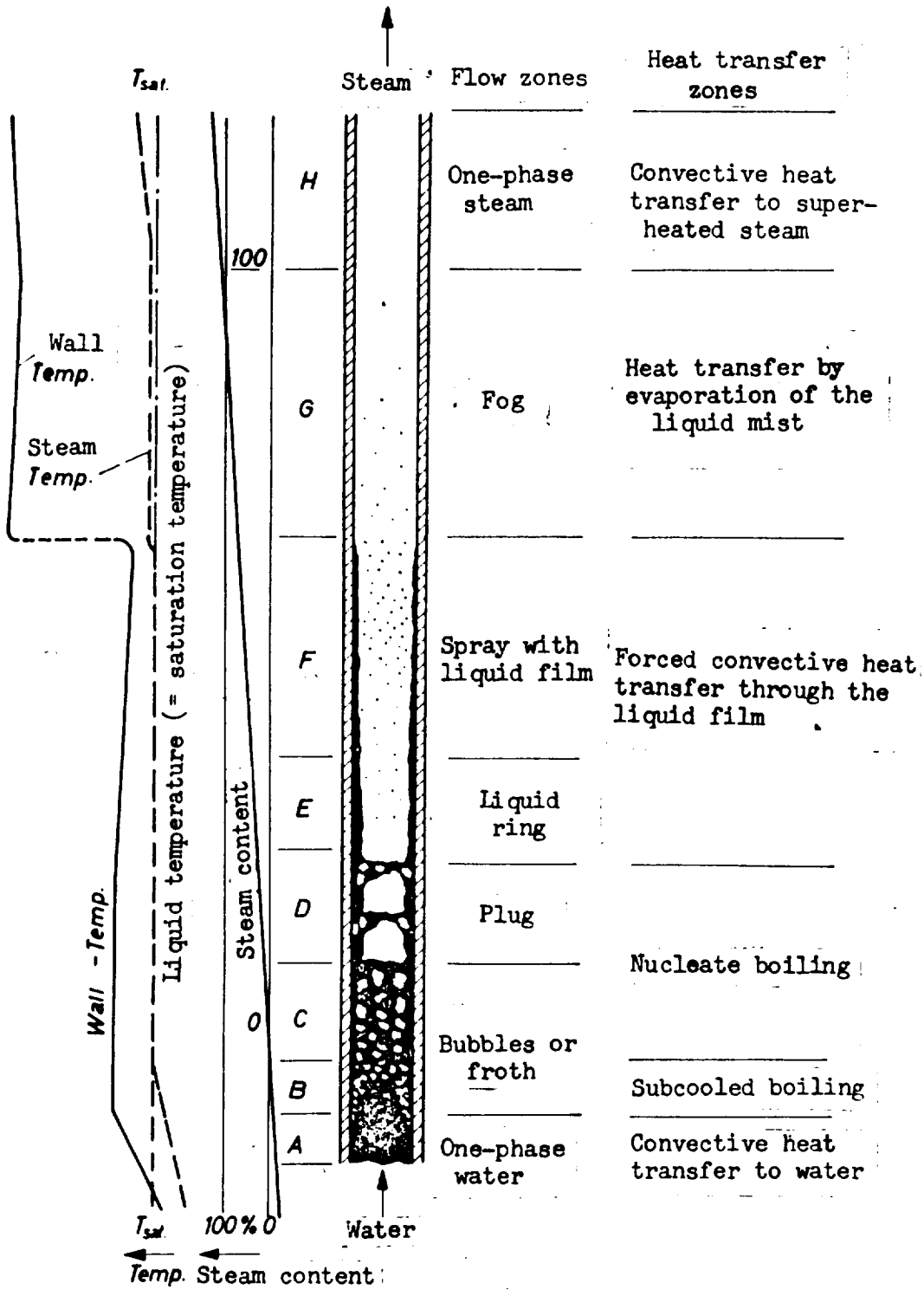


Fig.8 Two-Phase Flow Regions according to Collier (Bibl.19)

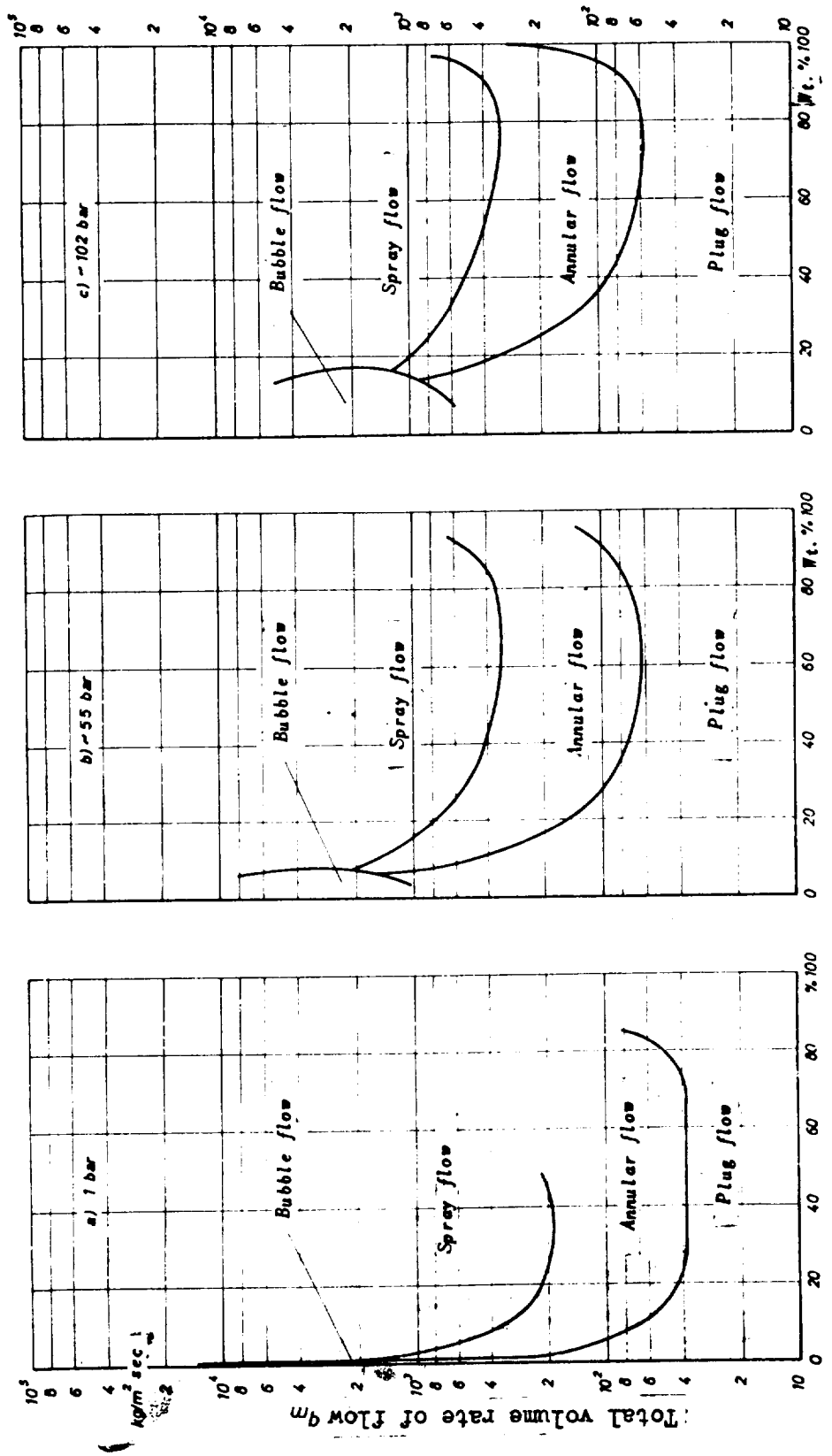


Fig.9 Flow Regions for Steam-Water at Various Pressures (Bibl.32)

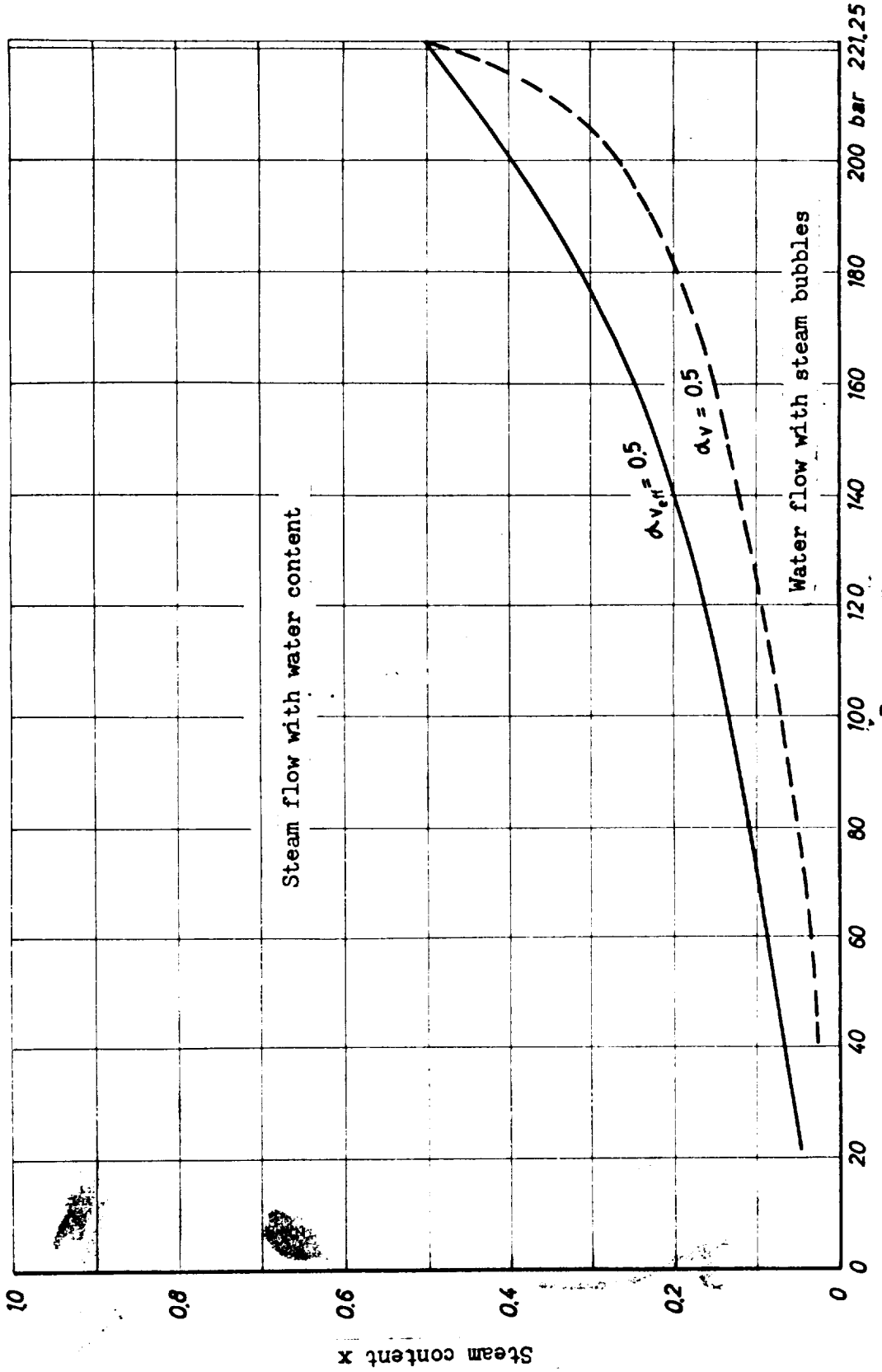


Fig.10 Maximum X-Values for Nucleate Boiling, as a Function of Pressure

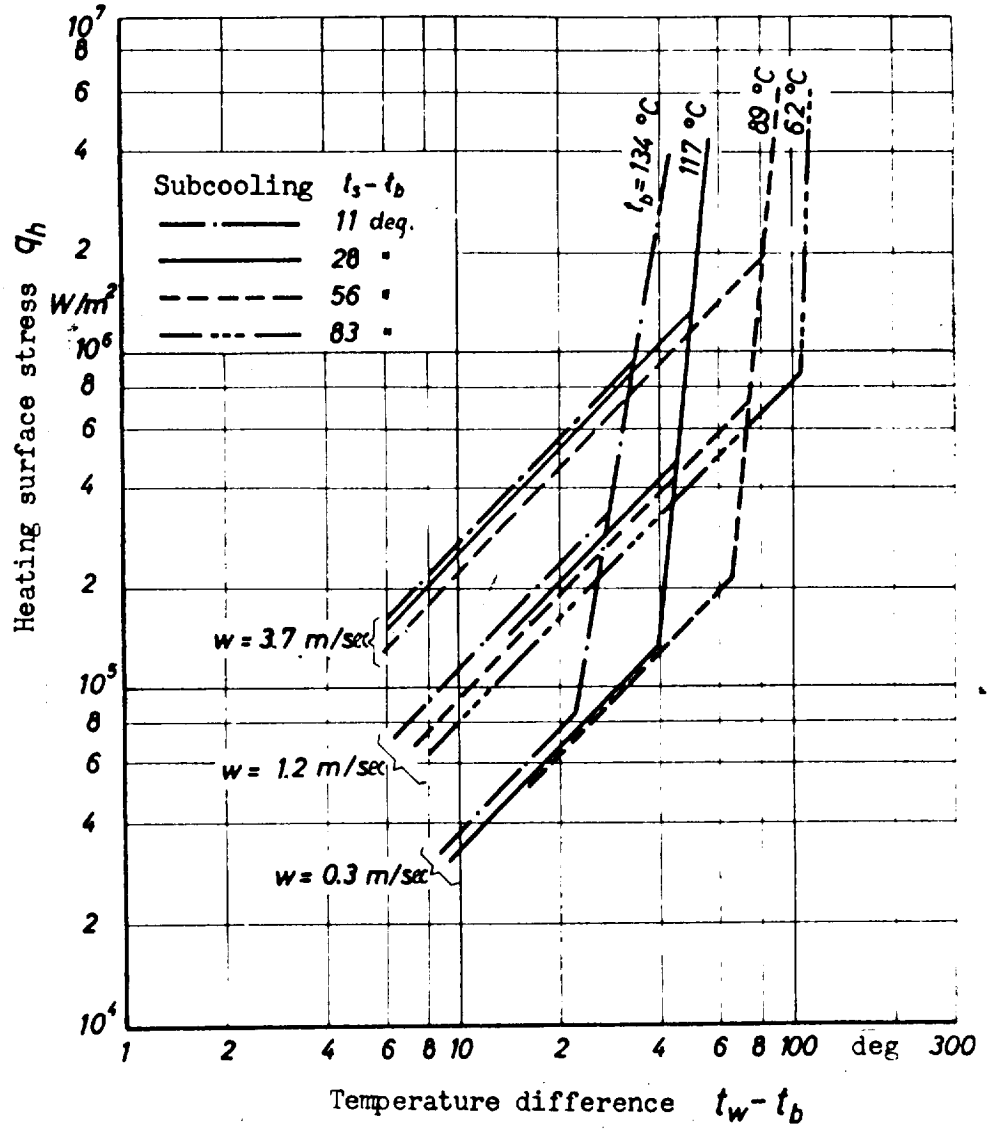


Fig.11 Influence of Subcooling and Velocity on the Heat Transfer Conditions, in Evaporation of Subcooled Water (q_h = Dependence on $t_w - t_b$) (Bibl.58)

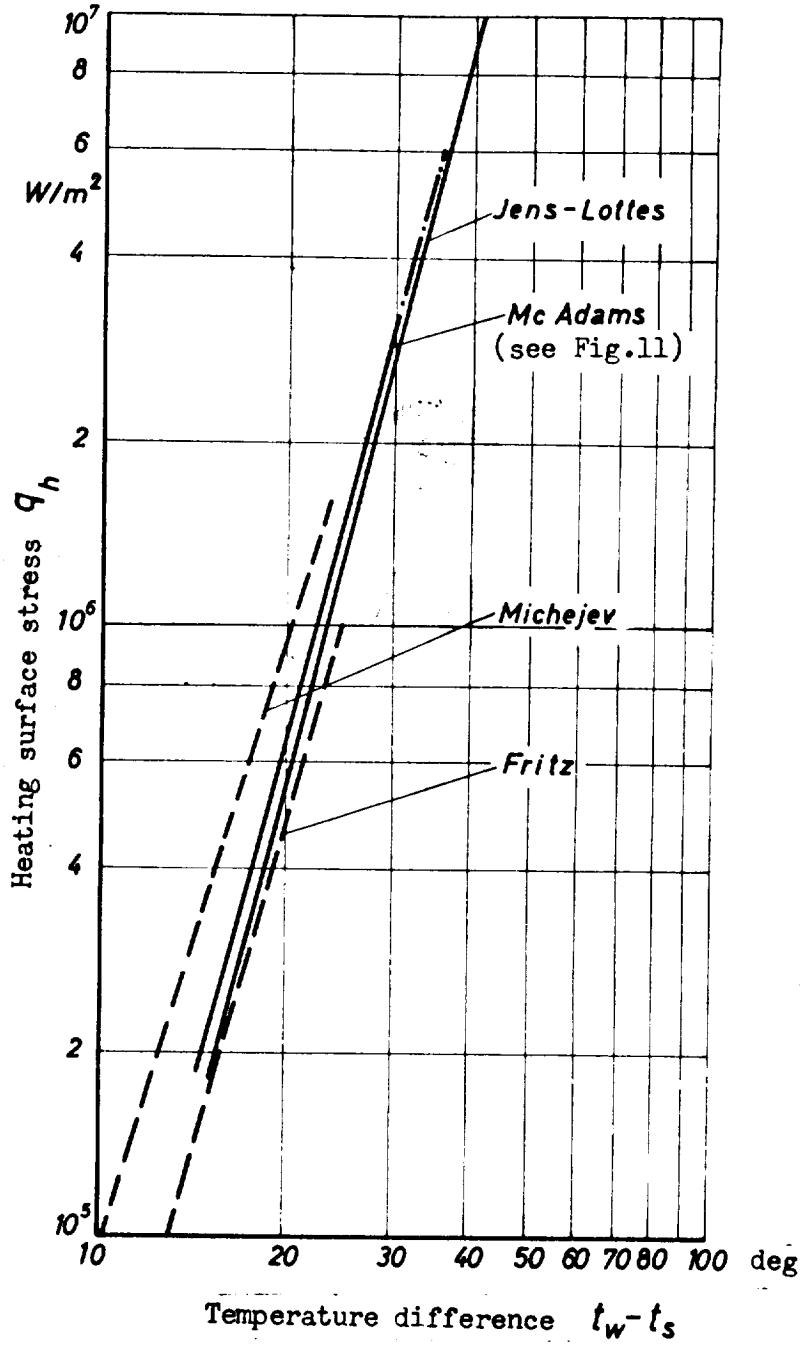


Fig.12 Comparison of Heat Transfer to Water, in the Region of Nucleate Boiling (q_h as a Function of $t_w - t_s$)

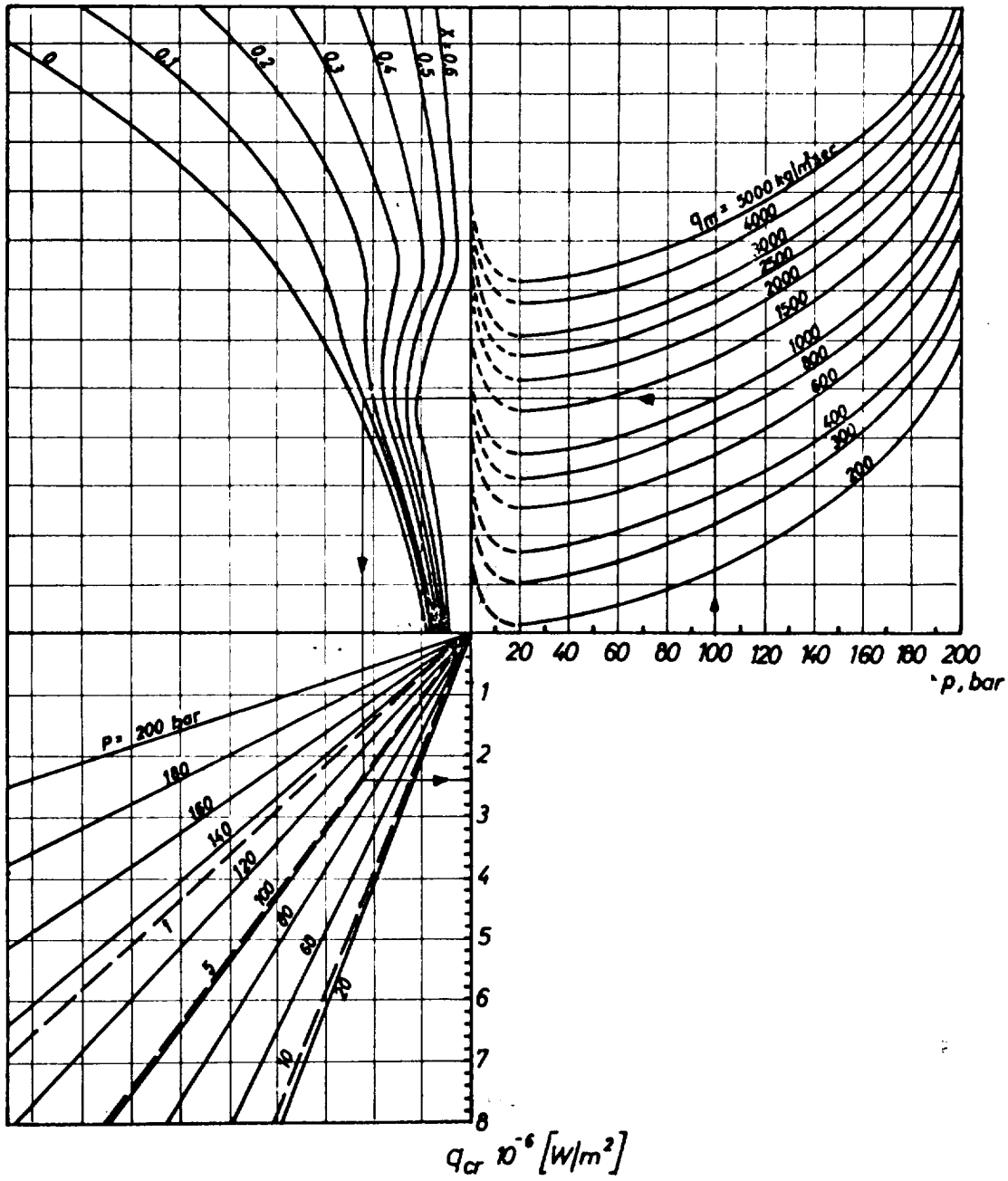
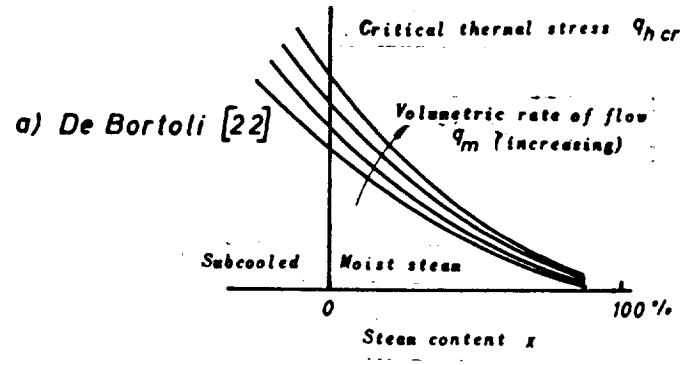
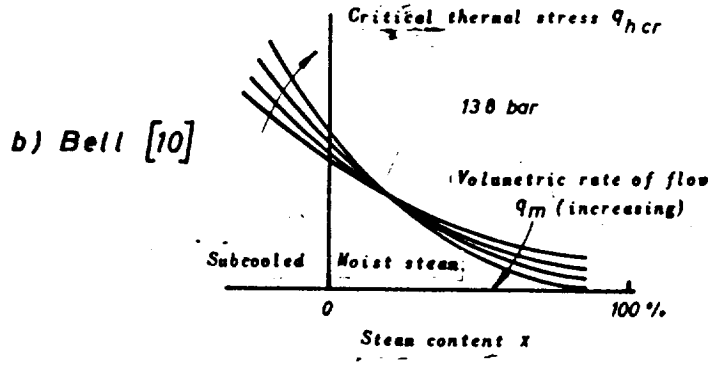


Fig.13 Nomogram for Determining the Critical Thermal Stress in Pipes and Annular Orifices ($\ell/d_h > 100$) (Bibl.61)





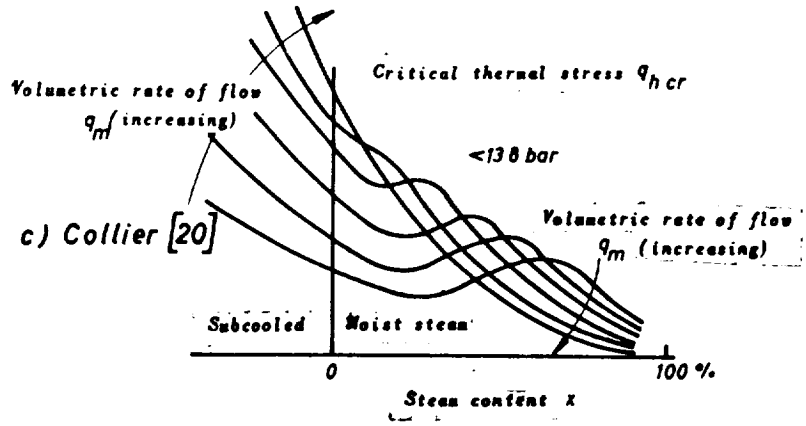


Fig.14 Slope of the Critical Thermal Stress as a Function of the Steam Factor (Bibl.20)

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